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Viscous Damping in the Drive Train of a Gas Turbine Engine

by T. A. Korjack

ARL-TR-1622

March 1998

Army Research Laboratory
Aberdeen Proving Ground, MD 21005-5067

ARL-TR-1622 **March 1998**

Viscous Damping in the Drive Train of a Gas Turbine Engine

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Abstract

Overdamping that is common to a single-degree-of-freedom damped linear vibratory system was extended to multidegree-of-freedom damped linear system, viz, a drive train situated in a typical gas turbine. Inequalities involving the mass, damping, and stiffness parameters were derived to form a system with a free response that was overdamped in each respective node. A general method to be employed in establishing the design parameters for designing systems to be overdamped in each mode has been identified for purposes of analysis, and the method was utilized to a four-degree- of-freedom model of a drive train in a gas turbine engine with a new look at a solution methodology for overdamping considerations. This technique, or method, for eliminating oscillations in n-degree-of-freedom lumped parameter systems by increasing the amount of viscous damping in the system has been illustrated by using actual data.

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1. Introduction

Viscous damping can limit the oscillations in mechanical systems such as in drive trains of modern tanks. If we look at a linear one-degree-of-freedom spring, mass, and dashpot arrangement, the selection of the proper values of mass, stiffness and damping constants to produce an overdamped or critically damped system can be effectuated without too much difficulty. The solution of a constant coefficient second-order ordinary differential equation shows that if $c \geq 2\,mk$ (where m, c, and k are the mass, damping, and stiffness coefficients, respectively) then the mechanical system will not oscillate. The purpose of this treatise is to show similar inequalities for nonoscillation of multidegree-of-freedom systems, especially inherent in the accessory gear box, reduction gear box and output shaft of a typical gas turbine engine. This type of analysis can prove to be invaluable for the diagnostics and prognostics of engines as illustrated by Helfman, Dumer and Hanratty (1995).

The systems considered here are those that can be modeled by the matrix differential equation

$$M\ddot{x}(t) + C\dot{x}(t) + Kx(t) = 0, \tag{1}$$

where x (t) is an n-dimensional vector of displacements and M, C, and K are nxn symmetric matrices containing the physical parameters of mass, damping, and stiffness constants. It is further assumed that M and K are positive definite matrices, and that C is at least a positive semi-definite matrix. The procedure presented in this work will take advantage of the derived matrix conditions (Inman and Andry 1980), to generate nonlinear algebraic inequalities for the physical parameters of the system. If the parameters can be chosen to satisfy these inequalities, the resulting transient energy response will be overdamped in each respective mode. The inequalities are stated explicitly in terms of the mass, damping, and stiffness constants of the system.

The exact relations for overdamping will be derived for a general two-degree-of-freedom system. As soon as the damping criterion becomes satisfied, the results will be used to calculate the

eigenvalues of the system to illustrate that the damped system is, in fact, overdamped in each mode. The design of a specific four-degree-of-freedom model of a drive train in a gas turbine engine asscociated with the reduction gear drive train in a tank will be given to illustrate the problems encountered in more practical design situations. The generalization to n degrees of freedom will become obvious from these examples.

2. Conception

Inman and Andry (1980) tell us that if, in addition to the restrictions previously listed, the matrices M, C, and K are such, that the matrix

$$M^{-1/2}\,CM^{-1/2} - 2\left(M^{-1/2}\,KM^{-1/2}\right)^{1/2} \tag{2}$$

is positive definite, then all of the eigenvalues of equation (1) will be negative real numbers, and hence each mode will be classified as overdamped. Since M is symmetric and positive definite, it possesses a unique positive definite square root, $M^{1/2}$ with inverse $M^{-1/2}$. Using the transformation

$$x = M^{-1/2}\,y,$$

equation (1) is reduced to

$$\ddot{y} + A\dot{y} + By = 0, \tag{3}$$

where $A = M^{-1/2}\,CM^{-1/2}$, and $B = M^{-1/2}\,KM^{-1/2}$. The condition for overdamping in each mode for equation (2) is that the matrix $A-2B^{1/2}$ must be positive definite. Since the square root of a matrix is, in general, harder to compute then the square of a matrix, it is tempting to use the matrix A^2-4B

in design work. Fortunately, it has been shown (Bellman 1968) that if A^2-4B is positive definite, then so is $A-2B^{1/2}$. Thus, requiring the matrix A^2-4B to be positive definite ensures that each mode of equation (2) will be a decaying nonoscillating (overdamped) function of time.

If it is desired to make the solution of equation (3) overdamped in each mode for arbitrary initial conditions, then it suffices to choose the physical constants m_i, c_i, and k_i so that A^2-4B is positive definite.

3. Methodology

To illustrate the previously mentioned ideas, consider the two-mass arrangement in Figure 1. The appropriate matrices for the equations of motion are

$$M = \begin{bmatrix} m_1 & 0 \\ 0 & m_2 \end{bmatrix}, \quad C = \begin{bmatrix} c_1 + c_2 & -c_2 \\ -c_2 & c_2 \end{bmatrix}, \quad \text{and} \quad K = \begin{bmatrix} k_1 + k_2 & -k_2 \\ -k_2 & k_2 \end{bmatrix}.$$

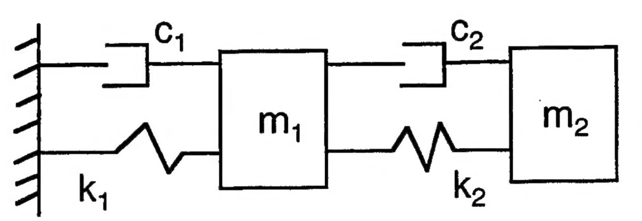

Figure 1. Two-Degree-of-Freedom System.

The matrix $M^{-1/2}$ is

$$M^{-1/2} = \begin{bmatrix} 1/\sqrt{m_1} & 0 \\ 0 & 1/\sqrt{m_2} \end{bmatrix}. \tag{4}$$

Denoting the i–jth element of a generic matrix A by A_{ij} and forming the matrix A^2-4B yields

$$(A^2 - 4B)_{11} = \frac{(c_1 + c_2)^2}{m_1^2} + \frac{c_2^2}{m_1 m_2} - 4\frac{k_1 + k_2}{m_1} \tag{5}$$

$$(A^2 - 4B)_{12} = \frac{-c_1 c_2 - c_2^2}{m_1\sqrt{m_1 m_2}} - \frac{c_2^2}{m_2\sqrt{m_1 m_2}} + \frac{4k_2}{\sqrt{m_1 m_2}} = (A^2 - 4B)_{21} \tag{6}$$

$$(A^2-4B)_{22} = \frac{c_2^2}{m_2^2} + \frac{c_2^2}{m_1 m_2} - 4\frac{k_2}{m_2}. \tag{7}$$

It is desired to choose m_i, c_i, and k_i so that the matrix A^2-4B is positive definite. A necessary and sufficient condition for a matrix D to be positive definite is for each of its leading principal minors to be positive. In particular, a real 2 × 2 matrix D is positive definite, if and only if

$$D_{11}>0, \quad \text{and} \quad D_{11}D_{22} - D_{12}D_{21} > 0 .$$

Applying these inequalities to $(A^2 - 4B)$ yields

$$\frac{(c_1 + c_2)^2}{m_1^2} + \frac{c_2^2}{m_1 m_2} > 4\frac{k_1 + k_2}{m_1} \tag{8}$$

$$\left[\frac{(c_1 + c_2)^2}{m_1^2} + \frac{c_2^2}{m_1 m_2} - 4\frac{k_1 + k_2}{m_1}\right] \times$$

$$\left[\frac{c_2^2}{m_2^2} + \frac{c_2^2}{m_1 m_2} - 4\frac{k_2}{m_2}\right] >$$

$$\left(\frac{1}{\sqrt{m_1 m_2}} 4k_2 - \frac{c_1 c_2 + c_2^2}{m_1\sqrt{m_1 m_2}} - \frac{c_2^2}{m_2\sqrt{m_1 m_2}}\right)^2 . \tag{9}$$

If the parameters m_i, c_i, and k_i are now chosen to satisfy equation (9) (along with the physical constraints that m_i, c_i, and k_i are all positive) then equation (1) will be overdamped in each mode and will not oscillate when perturbed from equilibrium. In total, the six parameters must satisfy eight inequalities (Brent 1973; Byrne and Hall 1973).

The approach taken here was simply to fix the values of m_i and k_i and chose values of c_i to satisfy equation (3). For example, the values

$$m_1 = 1 \qquad m_2 = 1$$
$$c_1 = 4 \qquad c_2 = 5$$
$$k_1 = 1 \qquad k_2 = 2$$

satisfy equation (9). In order to verify that this set of values implies overdamping, we solve the eigenvalue problem using these parameters. This yields the characteristic polynomial

$$\lambda^4 + 14\lambda^3 + 25\lambda^2 + 13\lambda + 2 = 0 \tag{10}$$

with eigenvalues

$$\lambda_1 = -0.2891 ,$$

$$\lambda_2 = -0.4652 ,$$

$$\lambda_3 = -1.2389,$$

and

$$\lambda_4 = -12.0068.$$

Thus, the design procedure yields an overdamped response, since each eigenvalue is a negative real number.

4. Design Application

The process described here may be useful in enhancing the survivability of certain structures by designing them to have an overdamped free response. In order to illustrate this in a design contest, we consider the drive train of a gas turbine engine (see Figure 2). The numerical values for inertia and stiffness are listed in the appendix, along with the definition of each parameter.

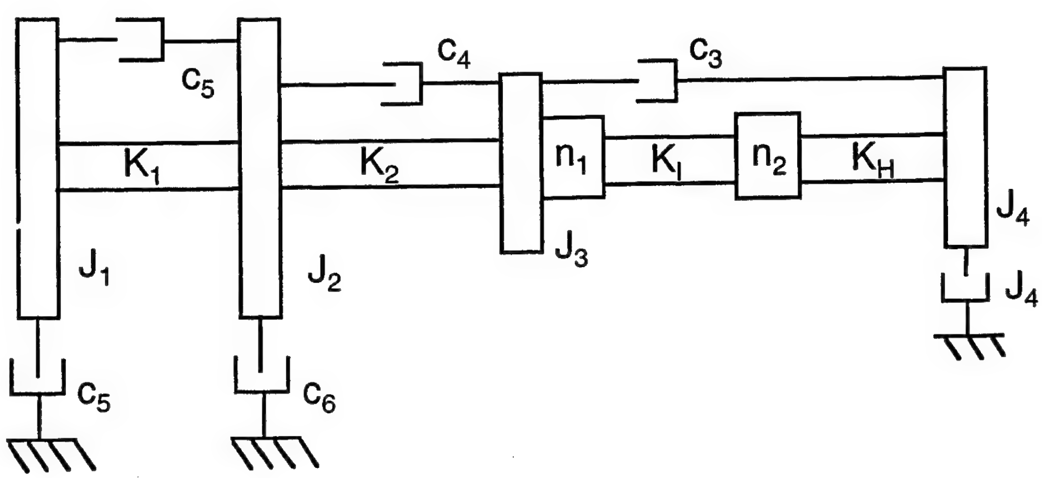

Figure 2. Schematic of Turbine and Drive Train Components.

In order to produce a C matrix that would allow inequalities similar to equation (9) to be formulated, some mechanism must be available for adding damping to the system. For non-

rotational systems, this may be accomplished by the use of shock absorbers or linear actuators. For dampers, this may be useful. Figure 2 indicates the addition of such dampers to an existing system (i.e., c_1, c_2, c_3, c_5, and c_6).

$$M = \begin{bmatrix} J_1 & 0 & 0 & 0 \\ 0 & J_2 & 0 & 0 \\ 0 & 0 & J_3 & 0 \\ 0 & 0 & 0 & J_4 \end{bmatrix}, \tag{11}$$

$$C = \begin{bmatrix} c_1 & -c_1 & 0 & 0 \\ -c_1 & c_1 + c_2 & -c_2 & 0 \\ 0 & -c_2 & c_2 + c_3 & -c_3 \\ 0 & 0 & -c_3 & c_3 + d_4 \end{bmatrix}, \tag{12}$$

and

$$K = \begin{bmatrix} k_1 & -k_1 & 0 & 0 \\ -k_1 & k_1 + k_2 & -k_2 & 0 \\ 0 & -k_2 & k_2 + k_3 & -k_4 n_1 n_2 \\ 0 & 0 & -k_4 n_1 n_2 & k_4 \end{bmatrix}, \tag{13}$$

where J_i = the various values of inertia; c_i = added damping constants; d_4 = the damping constant, due to a possible connection strut; k_1 and k_2 = shaft stiffness constants; and k_3 and k_4 = stiffness constants associated with the transmission and gear system. The transmission has a gear ratio n_1, and the timing gear has a ratio of n_2. Forming the matrix $A^2 - 4B$ yields

$$(A^2 - 4B)_{11} = \frac{c_1 2}{J_1^2} + \frac{c_2^2}{J_1 J_2} - 4\frac{k_1}{J_1}, \tag{14}$$

$$(A^2 - 4B)_{12} = (A^2 - 4B)_{21} = -\frac{c_1^2}{J_1\sqrt{J_1J_2}} - \frac{c_1(c_1 + c_2)}{J_2\sqrt{J_1J_2}} + \frac{4k_1}{\sqrt{J_1J_2}}, \tag{15}$$

$$(A^2 - 4B)_{13} = (A^2 - 4B)_{31} = \frac{c_2^2}{J_2\sqrt{J_1J_3}}, \tag{16}$$

$$(A^2 - 4B)_{14} = (A^2 - 4B)_{41} = 0, \tag{17}$$

$$(A^2 - 4B)_{22} = \frac{c_1^2}{J_1J_2} + \frac{(c_1 + c_2)^2}{J_2^2} + \frac{c_2^2}{J_2J_3} - 4\frac{k_1 + k_2}{J_2}, \tag{18}$$

$$(A^2 - 4B)_{23} = -\frac{c_2(c_1 + c_2)}{J_2\sqrt{J_2J_3}} - \frac{c_2(c_2 + c_3)}{J_3\sqrt{J_2J_3}} + 4\frac{k_2}{\sqrt{J_2J_3}}$$

$$= (A^2 - 4B)_{32}, \tag{19}$$

$$(A^2 - 4B)_{24} = (A^2 - 4B)_{42} = \frac{c_2c_3}{J_3\sqrt{J_2J_4}}, \tag{20}$$

$$(A^2 - 4B)_{33} = \frac{c_2^2}{J_2J_3} + \frac{(c_2 + c_3)^2}{J_3^2} + \frac{c_3^2}{J_3J_4} - 4\frac{k_2 + k_3}{J_3}, \tag{21}$$

$$(A^2 - 4B)_{34} = -\frac{c_3(c_2 + c_3)}{J_3\sqrt{J_3J_4}} - \frac{c_3(c_3 + d_4)}{J_4\sqrt{J_3J_4}} + 4\frac{n_1n_2k_4}{\sqrt{J_3J_4}}$$

$$= (A^2 - 4B)_{43}, \tag{22}$$

and

$$(A^2 - 4B)_{44} = \frac{c_3^2}{J_3 J_4} + \frac{(c_3 + d_4)^2}{J_4^2} - 4\frac{k_4}{J_4}. \tag{23}$$

The addition of c_1, c_2, c_3, c_5, and c_6 is necessary to make A^2-4B positive definite. Requiring the four leading principal minors of the 4×4 matrix A^2-4B to be positive yields four inequalities in the inertia, damping and stiffness parameters. Using the values for J_i and k_i (listed in the appendix) and choosing c_i to satisfy the inequalities yields

$$c_1 = 12.0000 \times 10^4 \quad \text{N-m} \quad \text{s/rad},$$
$$c_2 = 5.8653 \times 10^4 \quad \text{N-m} \quad \text{s/rad},$$
$$c_3 = 1.4700 \times 10^2 \quad \text{N-m} \quad \text{s/rad},$$
$$c_4 = 3.5300 \times 10^2 \quad \text{N-m} \quad \text{s/rad},$$
$$c_5 = 18.0000 \times 10^4 \quad \text{N-m} \quad \text{s/rad},$$

and

$$c_6 = 7.1347 \times 10^4 \quad \text{N-m} \quad \text{s/rad}$$

as one possible solution for the added damping constants.

The characteristic polynomial for this system is

$$2.024397 \times 10^{-5}\, \lambda^8 + 8.57279 \times 10^{-3}\, \lambda^7 + 1.186625\, \lambda^6$$

$$+ 58.521594\, \lambda^5 + 7.830627 \times 10^2\, \lambda^4$$

$$+ 2.5788556 \times 10^3\, \lambda^3 + 5.7545081 \times 10^3\, \lambda^2$$

$$+ 5.7354329 \times 10^3\, \lambda + 16.620975 = 0, \tag{24}$$

which has roots

$$\lambda_1 = -0.002,$$

$$\lambda_2 = -1.707,$$

$$\lambda_3 = -14.814,$$

$$\lambda_4 = -68.894,$$

$$\lambda_5 = -1.5 \times 10^2,$$

$$\lambda_6 = -1.85 \times 10^2,$$

$$\lambda_7 = -0.943 + 2.216\ i,$$

and

$$\lambda_8 = -0.943 - 2.216\ i$$

5. Conclusion

A method accompanied by the complexity of the process and its level of applicability has been presented for eliminating oscillations in n-degree-of-freedom lumped parameter systems by increasing the amount of viscous damping in a four-degree-of-freedom system.

Another method available to produce total overdamping is given in Beskos and Boley (1980) for two-degree-of-freedom systems. Unfortunately, to extend the process in Beskos and Boley (1980) to n degrees of freedom requires a closed-form solution of polynomials of degree (n − 1). However, the method here requires only the numerical solution of nonlinear equalities. In addition, for the two-degree-of-freedom case, the method presented in Bellman (1970) allows only the parameters c_1 and c_2 to be adjusted. As an alternative, the method presented here allows all of the parameters m_i, c_i, and k_i to be sensitized i.e., adjusted to effectuate mass, damping and stiffness characteristics. Hence, it seems quite apparent that this method is more advantageous for design work especially when it involves computationally intensive operations.

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Appendix:

Parameter Values

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The parameter values of the turbine for the case of tip speed (ratio of 2) and turbine rotational speed of 5,000 RPM are

$J_1 = J_2$ = 1/2 of turbine motor inertia = 1.65×10^4 N − s³ − m (1.46×10^5 lb − s² − in)

J_3 = transmission inertia = 2.43×10^2 N − s² − in (2.15×10^3 lb − s² − in)

J_4 = generator inertia = 3.06 N − s² − in (27.1 lb − s² − in)

n_1 = transmission gear ratio = 35.6

n_2 = pulley gear ratio = $\dfrac{178{,}000}{(35.6)\ (5{,}000)}$ = 0.999

k_1 = rotor stiffness = 1.65×10^5 N-m/rad (1.46×10^6 lb − in/rad)

k_2 = shaft stiffness = 2.69×10^5 N − m/rad (2.39×10^6 lb − in/rad)

K_I = transmission shaft stiffness = 1.41×10^5 N − m/rad (1.25×10^6 lb − in/rad)

K_H = generator shaft stiffness = 2.10×10^3 N − m/rad (1.86×10^4 lb − in/rad)

$$k_3 = \frac{n_1^2 n_2^3 K_I K_H}{k_1 + n_2^3 K_H} = 2.62 \times 10^6 \text{ N} - \text{m/rad } (2.39 \times 10^9 \text{ lb} - \text{in/rad})$$

$$k_4 = \frac{k_3}{n_1^2 n_2^3} = 2.07 \times 10^3 \text{n} - \text{m/rad } (1.83 \times 10^6 \text{ lb} - \text{in/rad})$$

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REPORT DOCUMENTATION PAGE

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1. AGENCY USE ONLY *(Leave blank)*	2. REPORT DATE	3. REPORT TYPE AND DATES COVERED
	March 1998	Final, April 97 - Nov 97

4. TITLE AND SUBTITLE

Viscous Damping in the Drive Train of a Gas Turbine Engine

5. FUNDING NUMBERS

4B010503350000

6. AUTHOR(S)

T. A. Korjack

7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES)

U.S. Army Research Laboratory
ATTN: AMSRL-IS-CI
Aberdeen Proving Ground, MD 21005-5067

8. PERFORMING ORGANIZATION REPORT NUMBER

ARL-TR-1622

9. SPONSORING/MONITORING AGENCY NAMES(S) AND ADDRESS(ES)

10. SPONSORING/MONITORING AGENCY REPORT NUMBER

11. SUPPLEMENTARY NOTES

12a. DISTRIBUTION/AVAILABILITY STATEMENT

Approved for public release; distribution is unlimited.

12b. DISTRIBUTION CODE

13. ABSTRACT *(Maximum 200 words)*

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14. SUBJECT TERMS

viscous damping; drive train; gas turbine engine.

15. NUMBER OF PAGES

20

16. PRICE CODE

17. SECURITY CLASSIFICATION OF REPORT	18. SECURITY CLASSIFICATION OF THIS PAGE	19. SECURITY CLASSIFICATION OF ABSTRACT	20. LIMITATION OF ABSTRACT
UNCLASSIFIED	UNCLASSIFIED	UNCLASSIFIED	UL

NSN 7540-01-280-5500

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